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HYDRAULIC LINEAR DRIVE, PARTICULARLY A HYDRAULIC TRANSMISSION ACTUATOR

The invention relates to a hydraulic linear drive, particularly a hydraulic transmission actuator, according to the characteristics of the preamble of Claim 1.

Hydraulic linear drives are used, for example, in the case of automated standard transmissions, for the synchronization of the transmission gears (see, for example, Johannes Loomann, "Zahnradgetriebe", 2nd Edition, Page 156, and on).

In the case of the hydraulic linear drives of the above-mentioned type, the dual piston bounded by two pressure chambers is in each case pushed toward the left or right as a result of correspondingly being acted upon by pressure. In many of the application cases, the two pressure chambers are sealed off by sealing elements arranged on the outer circumference of the piston.

Particularly in the case of hydraulic transmission actuators, high actuating forces are applied during the synchronization of the transmission gears, which require a

reliable and durable sealing-off or separation of the two pressure chambers.

It is therefore an object of the invention to improve the sealing-off of the two pressure chambers in the area of the piston unit. This object is achieved by means of the characteristics indicated in Claim 1.

As a result of the fact that the actuating piston is constructed in two parts and a sealing element is arranged between the two piston parts, the sealing element is clamped between the two piston parts when the piston unit is adjusted and, because of the actuating forces to be applied, for example, during the synchronization of the transmission gear, is pressed radially toward the outside to a certain extent, so that the sealing between the actuating piston and the interior cylinder wall is advantageously improved.

By means of the characteristics indicated in the subclaims, additional advantageous embodiments and further developments of the hydraulic linear drive can be obtained.

The sealing element constructed as a sealing ring is received on a sealing device carrier which is axially guided on one of the two piston parts.

For a better axial guidance of the sealing device carrier, the latter engages on the face in the first piston part.

The sealing device carrier is shaped in one piece out of one of the two piston parts or, as an alternative, is arranged as a separate component between the two piston parts.

The sealing device carrier is advantageously longitudinally displaceably disposed on the first piston part, for limiting the contact pressure force exercised upon the sealing ring, the relative movement of the sealing device carrier being limited by two stops constructed on the first piston part.

An advantageous embodiment of a hydraulic linear drive which is adapted to the use as a hydraulic transmission actuator is obtained when the two piston parts and the cylinder housing have a stepped construction. As a result of the step piston which is forming in this manner, in a first adjusting path, a high adjusting speed can be achieved with a low friction while, because of a large piston diameter, a high actuating force can be generated about the synchronization point and thus a high radial contact pressure force of the sealing ring against the interior wall of the cylinder housing.

A longitudinal groove is formed in the surface area of the piston part section having a reduced diameter, which longitudinal groove in each case connects a first hydraulic chamber section with a second hydraulic chamber section of the two step pistons.

One control conduit respectively is connected to the two first hydraulic chamber sections of the two step pistons, which control conduit is used for the feeding or removal of hydraulic oil.

Two embodiments of the invention are illustrated in the drawing and will be described in detail in the following.

Figure 1 is a view of a liner drive with a hydraulic control according to a first embodiment; and

Figure 2 is a linear drive having a hydraulic control according to a second embodiment.

The hydraulic linear drive, which is illustrated in Figure 1 and can be used, for example, as a hydraulic transmission actuator for an automated standard transmission, has a two-part cylinder housing 2a and 2b which are both mutually connected, preferably screwed together, on their faces. In the cylinder space formed by the cylinder housing, two piston parts are

received which are called step pistons 4 and 6 in the following and, both being equipped with one piston rod 8 and 10 respectively guided out of the cylinder housing 2, are longitudinally displaceably guided in the cylinder housing 2. In this case, the sealing-off of the two pistons rods 8 and 10 takes place by means of one sealing ring 12 and 14 respectively. The two step pistons 4 and 6 each have two piston sections 4a, 4b and 6a, 6b respectively, in which case a sealing device carrier 18 with a sealing ring 16 is arranged between the mutually facing faces of the pistons section 4b and 6b provided with a larger diameter. The sealing device carrier 18 is disposed on an interior ring flange section 20 of the piston section 4b and, on its right face, is screwed to the piston section 6b of the step piston 6, while, on its left face, it engages by means of a ring flange 22 in a gearing manner in a ring groove 24 constructed between a central ring flange section 23 and an outer ring flange section 25 of the piston section 4b. The sealing ring 16 is pushed onto the ring flange 22 and correspondingly seals off the two pressure chambers 26 and 28 from one another which are separated by the step pistons 4 and 6.

For limiting the sealing device carrier 18 longitudinally displaceably disposed on the interior ring flange section 20, a left and a right stop is provided, the left stop being formed by the central ring flange section 23 of the piston section 4b, and

the right step 32 being formed by a limit stop washer 32a which is axially secured by a snap ring 32b received in a ring groove. Further, a flat coil spring 34 is arranged on the interior ring flange section 20, is accommodated in a ring groove forming between the interior and central ring flange section 20 and 23 and is therefore clamped in between the sealing device carrier 18 and the piston section 4b.

The two piston sections 4a and 6a respectively have a longitudinal groove 36 and 38 made in the surface area, which longitudinal groove 36 and 38 respectively hydraulically connects the pressure chamber 26 and 28 respectively with a second pressure chamber 40 and 42 respectively. The two pressure chambers 40 and 42, in the following called first pressure chambers, are bounded in each case by the face 41 and 42 respectively of the piston section 4a and 6a and the face of the sealing ring 12 and 14 respectively. One hydraulic conduit 44 and 46 respectively is connected to the two first pressure chambers 40 and 42, by way of which hydraulic conduit 44 and 46, by means of a control valve 48, these pressure chambers 40, 42 can optionally be supplied with hydraulic oil from a tank 50. One return flow conduit 49 and 51 is in each case connected to the two pressure chambers 26 and 28 respectively, in the following, called second pressure chambers, which return flow conduit 49 and 51 can optionally be connected by way of the

control valve 48 with the tank 50.

In the following, the method of operation of the hydraulic linear drive will be described in the following:

In the control position of the 7/2-way valve 48 illustrated in Figure 1, the first pressure chamber 42 is acted upon by hydraulic oil by way of the hydraulic conduit 46 for the displacement of the two step pistons 4 and 6 toward the left. By means of the actuating force exercised on the face 43 of the piston section 6a, the piston unit consisting of the two step pistons 4 and 6 is displaced toward the left, in which case, after a first adjusting path, by way of the longitudinal groove 38 connecting the two pressure chambers 42 and 28, the second pressure chamber 28 is also filled with hydraulic oil. After a further distance, the hydraulic oil arrives in an unthrottled manner from the first pressure chamber 42 in the second pressure chamber 28 and acts exclusively with respect to the piston section 6b with the larger diameter, so that, on the one hand, the adjusting rate of the actuating piston 4, 6 is reduced but, on the other hand, the actuating force acting upon the step piston 6 is increased. Simultaneously, the hydraulic oil situated in the first and second pressure chamber 40 and 26 of the opposite side is returned into the tank 50 by way of the return conduit 49 and the hydraulic conduit 44. The fact that

the piston unit 4, 6 is displaced against a resistance, has the effect that the sealing ring 16 clamped in between the exterior ring flange section 25 of the piston section 4b and the sealing device carrier 18 deforms elastically and is thereby pressed radially against the interior wall of the cylinder housing 2.

The hydraulic linear drive can be used, for example, as a hydraulic transmission actuator, in which case a shift fork engaging in a gearshift sleeve unit is axially displaced by means of the transmission actuator for establishing a non-rotatable connection between the gearshift sleeve and the transmission gear. In this case, a high adjusting speed with a low friction is reached by way of a first adjusting path by means of the two piston sections 4a and 6a respectively which have the smaller diameter, while about the synchronization point, a high radial contact pressure force of the sealing ring 16 can be achieved with respect to the interior cylinder wall by means of the two piston sections 4b and 6b respectively which have a larger diameter.

The second embodiment of the hydraulic linear drive illustrated in Figure 2 differs only with respect to the hydraulic control. Instead of the 7/2 control valve 48 used in the first embodiment, the controlling of the feeding and removal of hydraulic oil now takes place by way of a first 4/2 control

valve 56 and a second 3/2 control valve 58. By way of the first 4/2 control valve 56, the two first pressure chambers 40 and 42 respectively can optionally be acted upon by hydraulic oil, while the return of the hydraulic oil from the first two pressure chambers 26 and 28 respectively is controlled by the control valve 58. The difference with respect to the first embodiment consists of the fact that, by the respective closing of the conduit 49 and 51, the hydraulic oil to be returned from the respective second pressure chamber 26 and 28 into the tank 50 is returned by way of the longitudinal groove 36 and 38 respectively, the respective first pressure chamber 40 and 42 respectively and the hydraulic conduit 44 and 46 respectively. As a result, an additional damping of the adjusting movement can be achieved, particularly when reaching one of the two end positions of the actuating piston 4, 6.